Parametric Studies of an Automotive Air Conditioning System

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Abstract — **The work carried out is the performance modeling of an automotive air-conditioning system. The modeling consists in describing the characteristics of the various components of the system and the study of their characteristics. The components are the heat exchangers (evaporator and condenser), the expansion valve and the compressor. Suitable functional relations are used in the mathematical description of these components. The component sub-models are integrated to form a cycle simulation model. The methodology for the modeling of individual parts is presented along with the flow chart of complete computer code written for the purpose. Results are obtained for various parameter values of condenser inlet air temperature, condenser face velocity and the rotational speed of the compressor. A considerable drop in the performance of the cycle is observed when the refrigerant tends to fall into the two-phase region at the condenser outlet. The decrement in the performance is because of the insufficient heat transfer at the condenser.**

Keywords — **Air conditioning system, Brazed aluminum heat exchangers, Heat transfer coefficient, Heat transfer rate, Pressure drop, Two-phase region**.

I. Introduction

Most automotive air conditioning systems work on the vapour compression principle. The main components of an automotive air conditioning system are the same as those in other air conditioning systems, namely, the compressor, condenser, expansion valve and the evaporator. But because of space constraints and better performance needed in the automotive air conditioning system, the swash plate compressor and brazed aluminum flat tube heat exchangers are used.Except for the similarity in functionality, automotive air conditioning systems differ from conventional systems in many ways. The automobile is directly exposed to different kinds of weather; cold, mild, damp, rainy and hot. In addition, provision is to be made for heating (if necessary), and defogging and de-icing of glazings. The control of dust, smoke and odours are additional important factors. By far,the most critical conditions arise in very hot weather and in slow running when the air flow over the condenser and the compressor speed can become insufficient to provide the necessary cooling. Solar load and fenestration are the main contributions to cooling load. When the vehicle is parked in hot weather with the doors closed, the hot soak temperature can reach 65-70°C. Thus the automotive air conditioning system needs to provide the thermal comfort under highly transient operating conditions. Literature review reveals that a limited amount of work has been done for the evaluation of performance of an automotive air-conditioning. Among these studies, Jabardo et al. [1] and Brown et al. [2] are noteworthy. The paper by Wu et al. [3] focuses attention on the design of heat exchangers. In the present paper, a parametric study is conducted on a typical automotive air conditioning system. The working fluid in the system is HFC134a. The thermodynamic properties are evaluated using the relations suggested in Cleland [4]. The transport properties of the refrigerant are calculated by the method of Chung et al. [5]. Properties of moist air are calculated by considering moist air to be a mixture dry air and water vapour.

Nomenclature

c_p	Specific heat (kJ/kgK)
c_R	Heat capacity ratio
dP/dz	Pressure loss per unit length (Pa/m)
f	Friction factor
g	Acceleration due to gravity (m/s^2)
G	Mass velocity $(kg/m^2 s)$
h	Enthalpy (kJ/kg)
k	Thermal conductivity (W/mK)
M	Mass flow rate (kg/s)
p	Pressure (kPa)
P_r	Prandtl number
R_e	Reynolds number
R_p	Pressure ratio
RPM	Round per minute (minute^{-1})
T	Temperature (°C)
V_{dis}	Displacement volume (m^3)
x	Dryness fraction

Greek letters

α	Heat transfer coefficient (W/m^2K)
α	Void fraction

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ε_{cv}	Clearance factor
η	Efficiency
ρ	Density ()
Ω	Tube angle with horizontal (degree)
ψ	Heat transfer enhancement factor

SUBSCRIPTS

1	Inlet point
c	Condenser
D_h	Hydraulic diameter
e	Evaporator
fr	Frictional
i	Inside tube
if	Interface
l	Liquid part
ma	Moist-air
o	Outside tube
r	Refrigerant
rl	Liquid refrigerant
s	Isentropic
v	Volumetric, Vapour par

II. MATHEMATICAL MODELING

The simulation program is based upon steady state mathematical models of the components of the refrigeration cycle including the compressor and heat exchangers. The action of the thermostatic expansion valve is modeled as constant superheat at the exit of the evaporator.

A. Compressor Model

An open type swash plate compressor with constant displacement volume is considered and modeled using empirical correlations developed for refrigerant mass flow rate, volumetric efficiency, isentropic efficiency and enthalpy change in isentropic compression [4] from the available data. The volumetric efficiency of the compressor is the result of many parameters which are correlated in following manner:

$$\eta_v = f(V_{dis}, RPM)\{1 - \varepsilon_{cv}[R_p{}^{1/C_R} - 1]\}$$

Since the displacement volume, V_{dis} is not variable in the present case, the function f is dependent only upon the speed of compressor. A curve fit from the available data has been used for the function .

Based on the data taken from several automotive compressors, Brown et al. [2] suggested the following expression for isentropic efficiency for pressure ratio greater than two: $\eta_s = 0.9343 - 0.04478R_p$

B. Heat Exchangers

Brazed aluminum heat exchangers used, which have high performance louvered fins soldered with the tubes and flat multichannel tubes with small hydraulic diameters. The mathematical formulation of these heat exchangers is similar for both condenser and evaporator, with the only difference arisingin the total heat transfer rate calculation, because a thin layer of flowing water film is to be considered over the evaporator surface when the surface temperature is below the dew point temperature. The total heat transfer rate and pressure drop in the heat exchangers are calculated in an incremental manner. The values obtained from each increment are added up to get the final values required.

Inside tube calculation, i.e. refrigerant side calculation has been done for four regions, namely liquid region, two-phase region, liquid-deficient region and the superheated region. In case of evaporator, there is no liquid region because the flow enters the evaporator at a certain quality. Two phase region heat transfer coefficient for vertical tubes is calculated with Shah's correlation [6, 7], as given below:

$$\alpha_r = \psi \alpha_{rl}$$

where, ψ is the heat transfer enhancement factor which depends upon Convection number, Boiling number and Froude number of liquid refrigerant and h_{rl} is the heat transfer coefficient of the liquid refrigerant flowing alone in the tube. The pressure drop during flow for this is calculated from following equation [3]:

$$\frac{dP}{dz} = \left(\frac{dP}{dz}\right)_{fr} - [(1-x)\rho_l + \rho_v \alpha]g \sin\Omega - \frac{d}{dx}\left[\frac{G^2 x^2}{\alpha \rho_v} + \frac{G^2(1-x)^2}{\rho_l(1-\alpha)}\right]$$

where, the first, second and the last terms denote respectively the effect of friction, gravity and acceleration or deceleration caused by the phase change. The heat transfer coefficient and friction factor in the superheated region are calculated using Petukhov [8] equation for turbulent flow as follows:

$$h_r = \frac{(f/2)Re_{D_h}Pr_v}{1.07 + 12.7(f/2)^{1/2}(Pr_v^{2/3} - 1)}\left(\frac{k_v}{D_h}\right)$$

$$f = [1.58\ln(Re_{D_h}) - 3.28]^{-2}$$

In the liquid deficient region which is assumed to occur between the dry-out point and the saturated vapour state, the heat transfer coefficient is calculated as linearly interpolated value. The dry-out point is considered to occur at a quality of 0.8. For the considered geometry of heat exchanger, i.e., corrugated louver with triangular channel, calculation of heat transfer coefficient and pressure drop of moist air outside the tube is done using the correlations mentioned in Park and Jacobi [9]. Once the heat transfer coefficients of refrigerant and moist air are calculated, theheat balance between refrigerant and tube in case of condenser and a heat balance between refrigerant and water film in case of evaporator can be made to estimate the incremental heat transfer rate. The heat balance equation for the condenser is obtained by the simple approach but for the case of evaporator, concept of cooling and dehumidification coil is used with Lewis number equal to unity, which gives the equation as shown below:

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$$\frac{T_{if}-T_{r,m}}{h-h_{if}}=\frac{1}{C_{P,ma}}\frac{dA_o}{dA_i}\frac{\alpha_o}{\alpha_r}$$

where, h_{if} is the saturation enthalpy of the interface i.e. water film present over the evaporator and depends only upon the interface temperature, T_{if}. After calculation of total heat transfer rate through the heat exchangers, the parameters of moist air leaving the heat exchangers are calculated by applying mass balance and energy balance in the control volume, which involves inlet air, heat exchanger and outlet air.

C. Thermostatic Expansion Valve

Since, the opening of the valve is the function of the superheat at the evaporator outlet the mass flow rate is taken proportional to the square root of pressure difference between condenser and evaporator and to the pressure difference between the bulb and evaporator as given below:

$$M_r = C(p_b - p_e - p_{sp})\sqrt{\rho_1(p_c - p_e)}$$

where, p_b, p_e and p_c are the bulb, evaporator and condenser pressure respectively. The constant C, presented in the equation depends upon the geometry of the valve and the spring pressure (also known as static pressure) p_{sp} depends upon the initial setting of the valve to fix the minimum superheat at the evaporator outlet, where the bulb pressure depends upon the superheat at the evaporator outlet. However, in this study, the effect of the thermostatic expansion valve is taken as a specified degree of superheat at the exit of the evaporator.

D. Cycle Simulation

After modeling of the four parts of the system separately, cycle simulation is done by integrating the individual parts and running the code with number of iterations until the values converge to a specified level. Two evaporators served by individual TEVs are considered for the frontal and rear portions of the vehicle.

The evaporator pressure is initially guessed for given operating conditions and it keeps updating until the mass flow rate in the system converges and a fixed superheat of 10°C at the evaporator exit is obtained. The flowchart of the computer code written for this purpose is presented in the Appendix.

III. Results and Discussions

Based on the mathematical formulation and simulation method presented in the previous sections, results are obtained for the analysis of performance of the system running at its steady state. Constant values of evaporator air inlet velocity and temperature as 2 m/s and 27°C are assumed for the calculation.The set of results shown in Fig. 2 presents the performance characteristics of the system at constant condenser inlet air velocity of 7 m/s. Figs. 1(a) to 1(c) show the characteriscs of the system for three different compressor speeds of 2000 RPM, 2500 RPM and 3000 RPM and the fourth graph (Fig. 1(d)) gives the outlet conditions of refrigerant at condenser outlet in form of subcooled temperature for these compressor speeds.

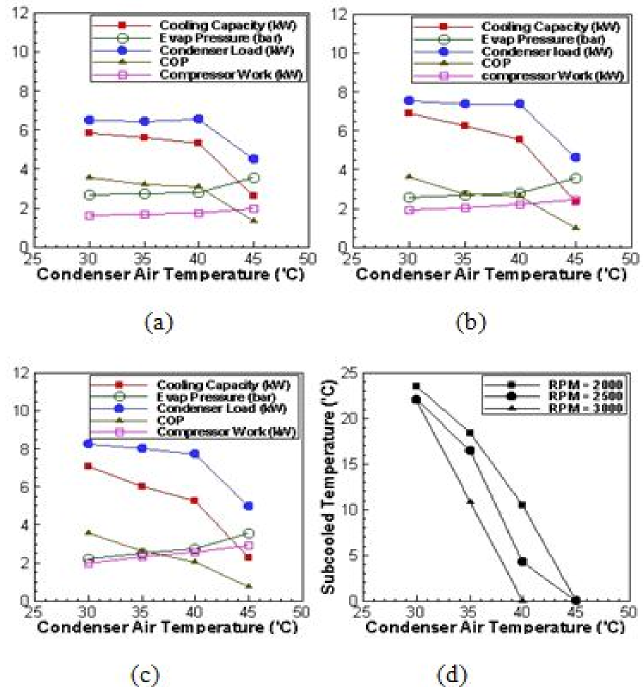

Figure 1. Cycle performance curves for condenser air velocity of 7m/sec at various air temperatures for (a). RPM = 2000, (b). RPM = 2500, (c). RPM = 3000 and (d). Refrigerant outlet condition at condenser exit.

From Fig. 1(d), it can be seen that curves are falling into two-phase region for higher air temperature. As compressor speed increases, the refrigerant mass flow rate increases. This makes the condenser exit state to fall into two-phase region at 40°C itself. From Figs. 1(a) to 1(c), it can be seen that the evaporator pressure and compressor work are increasing with increase in air temperature but other parameters, i.e. cooling capacity, condenser heat transfer and COP of the cycle are decreasing with sudden change in the slope of the curves when the refrigerant enters into two-phase region. The graphs show the degradation of system performance with increase in air temperature.The set of results shown in Fig. 2 presents the system performance at constant condenser inlet air temperature of 35°C. It can be seen that the performance of the cycle improves continuously for increase in the air velocity, i.e., vehicle speed. Fig. 2(d) shows the falling of refrigerant into two-phase region for air velocity less than 3 m/s, in case of 3000 RPM when refrigerant mass flow rate is high, which indicates the requirement of an external fan when vehicle speed crosses a minimum value for the respective air temperatures.

Conclusions

An integrated model, consisting of the sub-models for the heat exchangers, compressor and the expansion valve is formulated for the simulation of an automotive air conditioning system. The model can be used as a design tool to predict

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the performance of the system in advance. For example, for the parameters chosen in the present study, the results show the requirement of an increased air velocity over the condenser for high ambient temperatures. Alternatively, the condenser should be redesigned to provided extra heat transfer area if it has to function at higher ambient temperatures.

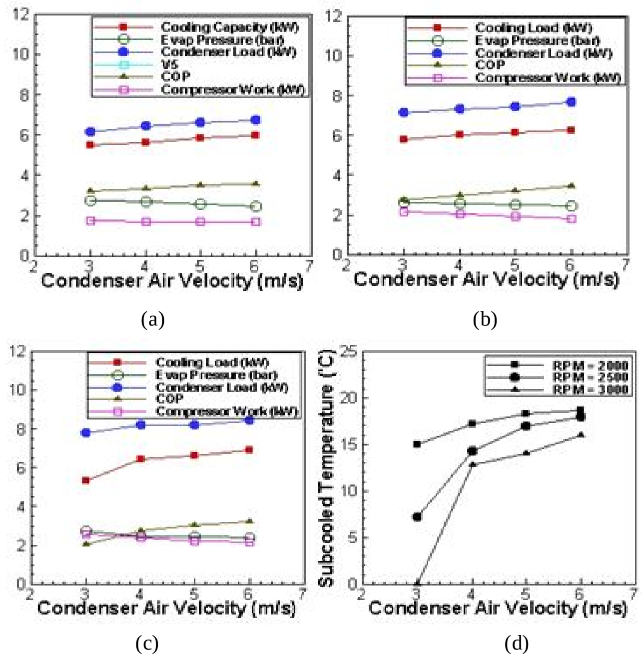

Figure 2. Cycle performance curves for condenser air tempe- rature of 35ºC at various air velocities for (a). RPM = 2000, (b). RPM = 2500, (c). RPM = 3000 and (d). Refrigerant outlet condition at condenser exit.

For efficient and smooth performance of an automotive air-conditioning system, the optimum design of all the four parts is crucial. The design and capacity of the components should match each other, so that the refrigerant will enter the components at their respective optimum thermodynamic state and system will reach its steady state soon. But, in a case if they don't support each other then system may lead to instability and it may get shut down.

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APPENDIX: Methodology presented in the form of flow chart

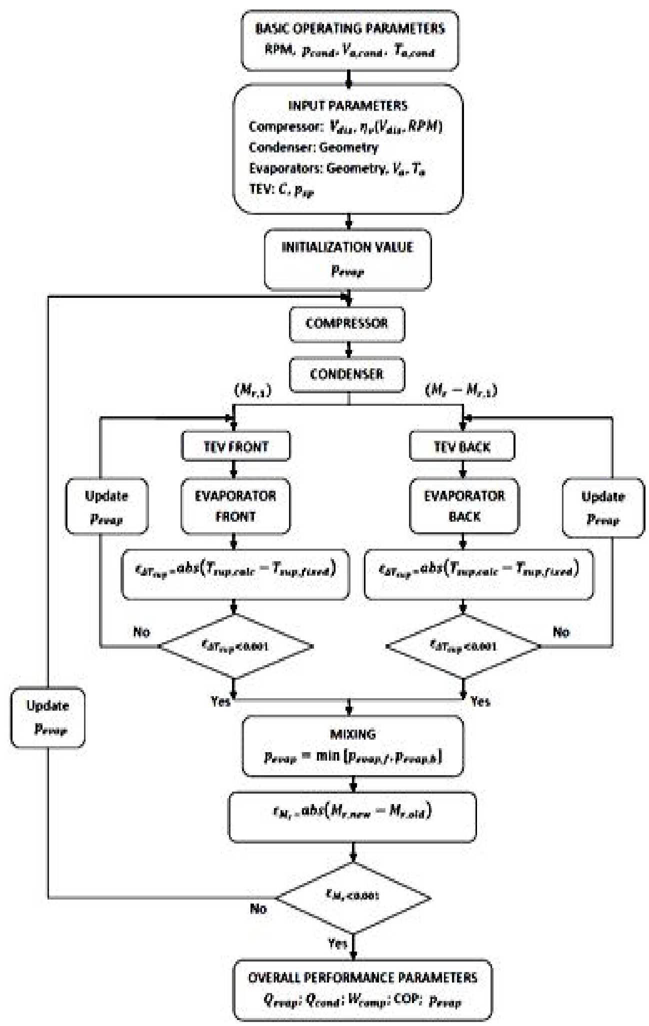

Figure 3. Flowchart of computer code for overall cycle simulation.

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